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**SHAFT FACE SEAL WITH
SELF-ACTING LIFT AUGMENTATION
FOR ADVANCED GAS TURBINE ENGINES**

by Robert L. Johnson and Lawrence P. Ludwig

Lewis Research Center

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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ABSTRACT

The need in advanced engines for shaft seals with improved pressure, temperature, and speed capabilities is pointed out. In particular, the high-speed requirement suggests that the seals must not have rubbing contact. Labyrinth seals have disadvantages in high leakage (efficiency loss) and debris entrainment, and conventional face seals are subject to high wear because face deformation can increase seal closing force and thus cause heavy rubbing contact. It is shown that incorporation of self-acting lift pads to the conventional face seal prevents rubbing contact even though face deformation occurs; feasibility was demonstrated in 560 hours of operation at 100-pound-per-square-inch (69-N/cm^2) pressure differential, 600°F (589 K) gas temperature, and 380-feet-per-second (116-m/sec) sliding speed. Positive separation of the sealing faces with acceptable leakage rates was obtained. Proposed advanced systems using seals with self-acting lift pads are described.

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SUMMARY

Shaft seals in advanced engines will operate at pressures, temperatures, and speeds higher than feasible with current face and shaft riding seals. The functions of the various shaft seals are discussed, and the debris entrainment and leakage rate disadvantages associated with labyrinth seals are pointed out. Speed and pressure limitations of conventional face seals are primarily due to seal face deformation. In particular, it is shown that divergent deformation of sealing faces leads to high-speed rubbing contact and, hence, high wear. High-speed rubbing can be avoided (seal face separation maintained) if self-acting lift pads are incorporated in the face seal. These pads augment seal lift and prevent rubbing contact even when face divergence occurs.

The feasibility of a seal with self-acting lift augmentation was demonstrated in both short term tests and in a run of 560 hours duration at 100-pound-per-square-inch (69-N/cm^2) pressure differential, 600° F (589 K) gas temperature and 380-feet-per-second (116-m/sec) sliding speed. Inspection of the seal faces after the tests indicated that the seals were running with positive separation of the faces. Additional successful operation in a short term test at 175-pound-per-square-inch (121-N/cm^2) pressure differential, 1000° to 1100° F (811 to 866 K) gas temperature and 400-feet-per-second (122-m/sec) sliding speed indicated higher operational capability than conventional face seals.

The low leakage rates obtained, together with the high-speed capability (due to positive separation of the face), suggest that the seal with self-acting lift augmentation is a candidate for advanced systems. Two proposed advanced seal systems using seals with self-acting lift pads are presented.

INTRODUCTION

Shaft seals are used in gas turbine engines to restrict leakage from a region of gas at high pressure to a region of gas at a lower pressure, and to restrict gas leakage into the bearing sumps. (Bearing sumps contain an oil-gas mixture at near ambient pressure, and gas leakage through the seal helps prevent oil leakage out and maintains a minimum sump pressure necessary for proper scavenging.) The shaft seals in advanced engines will be required to operate at pressures, temperatures; and sliding speeds greater than current practice. For example, some high-pressure ratio advanced engines are expected to operate to 500 pounds per square inch (345 N/cm^2) at the compressor discharge. Higher overall temperatures, which accompany increases in flight speed (ref. 1), add to the sealing problem by increasing seal operating temperatures; higher seal sliding speeds are partly due to larger diameter shafts, and speeds to 500 feet per second (152 m/sec) can be anticipated in some advanced engines (ref. 2).

Seal leakage affects overall engine efficiency. Some of the gas which leaks through seals can be used in cooling, thus, not all leakage is a total loss. However, at 500-pound-per-square-inch (345-N/cm^2) compressor discharge, the loss at some of the locations, such as the turbine cooling gas seal, will be significant if low leakage seals are not developed.

Gas leakage through the sump seals is necessarily low and, therefore, not significant from an efficiency standpoint. However, gas leakage into the sump is important because the magnitude of this leakage of hot gas affects lubricant life (degradation) and also affects the cooling requirement of the lube system. The cooling requirement is important in high-speed flight in which there is competition for the heat sink of the fuel. In addition to seal leakage, the heat generated at the seal sliding interface must be considered in the cooling requirement evaluation. Thus, minimizing seal rubbing contact is advantageous.

The objectives of this study are to: (1) review functions of shaft seal systems and point out seal requirements for advanced engines; (2) identify problem areas in conventional shaft seals and describe seal concept using self-acting lift pads to mitigate problems encountered in conventional face seal; and (3) present pertinent experimental results, obtained in various NASA contracts, on seals with the lift pads and show how these seals can be applied to advanced systems.

Experimental data of seals with self-acting lift pads was obtained in studies under NASA Contract Number NAS3-7609 (ref. 3). In these studies a 7.05-inch (17.90-cm) mean-diameter seal was operated in a simulated engine sump to 200-psi (138-N/cm^2) pressure differential.

Some of the experimental data presented herein was obtained at the Stein Seal Company under NASA Contract NAS3-12454 on a 7.05-inch (17.90-cm) mean-diameter face seal with self-acting lift augmentation (lift pads). Leakage rates were measured at pressure

differentials to 175 pounds per square inch (121 N/cm^2), at gas temperatures to 1100° F (866 K), and at seal sliding speeds to 400 feet per second (122 m/sec).

Other experimental data were obtained with a 6.23-inch (15.80-cm) mean-diameter face seal with self-acting lift pads operating in a simulated engine sump using an inerted lubrication system. These data, obtained under Contract NAS3-6267 (ref. 4), are for seal operation at 100-pound-per-square-inch (69 N/cm^2) pressure differential, 600° F (589 K) gas temperature, and 380-feet-per-second (116 m/sec) sliding speed.

Additional results of interest were obtained at AiResearch Manufacturing Co. under NASA Contract NAS3-9428 on a 1.60-inch (4.07-cm) mean-diameter face seal with self-acting lift pads operating at 250 feet per second (76 m/sec) with a small pressure difference of 2 pounds per square inch (1.4 N/cm^2) across the seal.

SYMBOLS

| | |
|-------|------------------------|
| b | width of flow passage |
| h | clearance, gap height |
| P | pressure |
| R | universal gas constant |
| r | radius |
| T | absolute temperature |
| W | weight flow |
| μ | absolute viscosity |

SEAL FUNCTIONS AND DESIGN CONSIDERATIONS

Current Seal Systems

Aircraft gas turbine engines contain a number of rolling element bearings which support the shaft(s), and which are enclosed by sumps for lubrication purposes. The number of bearings used varies, some engine types employ two, and others as many as eight. Figure 1, a schematic of a 3-bearing design, illustrates the mainshaft seals at the various bearing locations. The sump seals at the front of the compressor usually present no problem since the sump can be surrounded by compressor bleed air of moderate pressure and temperature. Bleed air leaks through the seal and, thus, pressurizes the front sump. For moderate bleed air pressures and temperatures a single shaft seal of

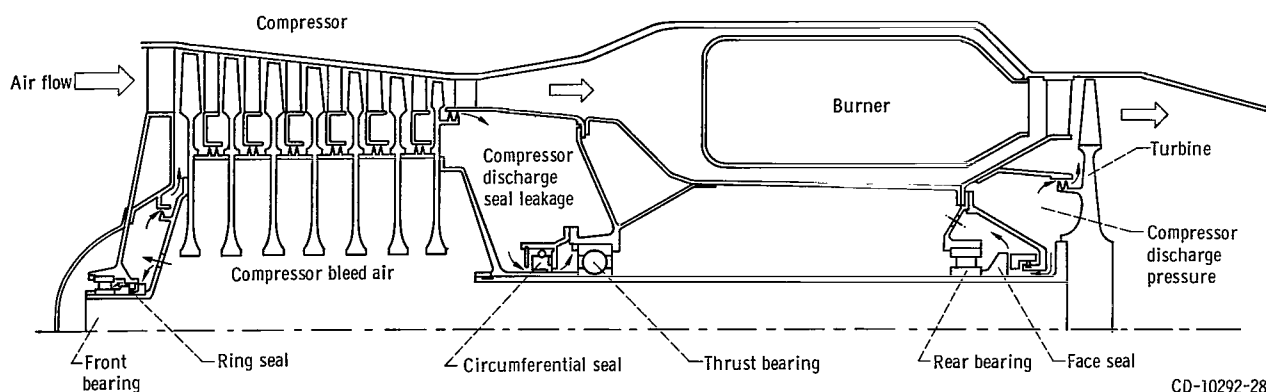


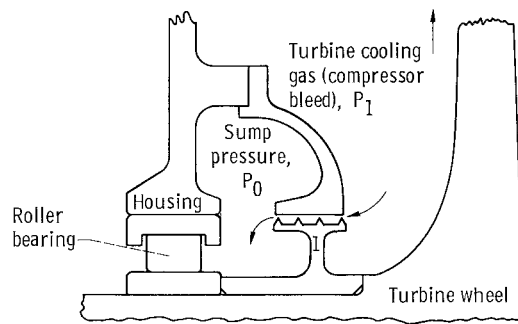
Figure 1. - Engine schematic showing seal locations.

many types is adequate. As an example, ring seals have been used at 54 pounds per square inch (37 N/cm^2) and 330° F (439 K); circumferential seals are more than adequate for this location, and in the present state of development, have been used to 85 pounds per square inch (58 N/cm^2), 700° F (644 K), and 240-feet-per-second (73-m/sec) sliding speed (ref. 5); face seals are also adequate as some are operating at 125 pounds per square inch (86 N/cm^2), 800° F (700 K), and 350-feet-per-second (107-m/sec) sliding speed (ref. 3). The final selection of the seal type for the front compressor location depends on the many design and system considerations, including preference.

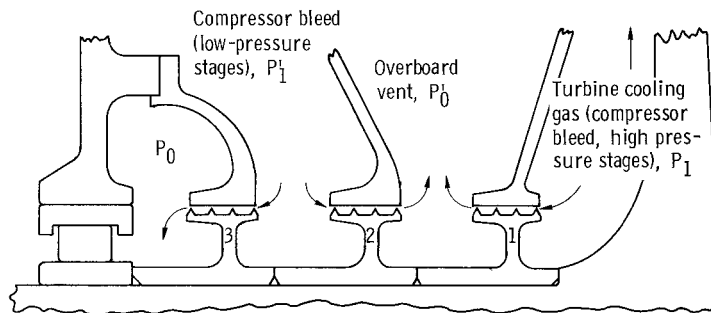
At the middle of the engine (fig. 1) and at the turbine bearing sump, the seal operational requirements becomes more severe. The bearing(s) and seals tend to be large because of shaft size, and because the thrust bearing must be large enough to carry the net thrust load on the rotating parts. In addition, the sump is usually surrounded by higher pressure gas (and corresponding high temperature) than at the front of the engine.

Figures 2(a) to (d) show shaft seal arrangements used at the turbine bearing sump location. Here the basic problem is protection of the bearing sump from the turbine cooling gas. In early engines the cooling gas pressure and temperature was relatively low and a single labyrinth seal, which restricted turbine cooling gas leakage into the sump, was adequate. At these pressures, the efficiency loss due to seal leakage was insignificant and no fire hazard existed since gas temperature was below the oil flash point. However, a disadvantage of the labyrinth seal, as compared to the close-clearance (ring, circumferential and face) seals, is easier passage of air borne water and dirt into the sump. In addition, for labyrinth seals, reverse pressure drops must be avoided to preclude high oil loss. Furthermore, ram air temperature (minimum temperature sump-pressurizing air available) increases with increasing flight speed. A flight speed limit exists, therefore, beyond which application of a labyrinth seal to the sump location poses a fire hazard.

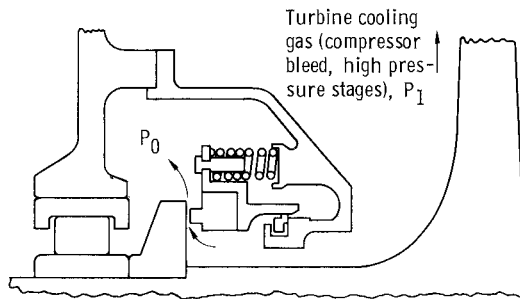
The required pressure of the turbine cooling gas is determined by the turbine inlet



(a) Single labyrinth, early engines.

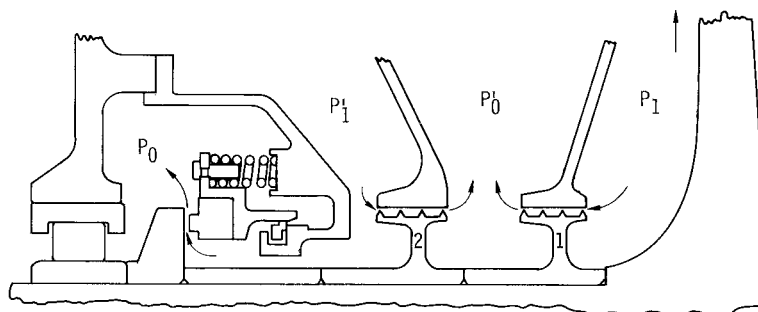


(b) Multiple labyrinth for high-temperature high-pressure turbine cooling gas.



(c) Conventional face seal.

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(d) Conventional face seal with labyrinth seal for high-temperature, high-pressure cooling gas.

Figure 2. - Seal systems (not to scale).

pressures. High-pressure compressor bleed or compressor discharge pressure is used to insure cooling flow radially outward along the turbine disk. As turbine cooling gas pressure requirements increased with engine development, the single labyrinth seal of figure 2(a) was no longer suitable, and the seal systems such as illustrated in figures 2(b) to (d) were used (no scale is intended by these schematic drawings). Figure 2(b) illustrates a seal system in which a labyrinth seal restricts the leakage of relatively high pressure (P_1), high-temperature turbine cooling gas to an overboard vent (P'_0) (or to a low-pressure region such as the tailpipe). Low-pressure compressor bleed (P'_1) surrounds the sump, hence, provides thermal protection, and leakage (through the labyrinth) into the sump provides required sump pressurization. The labyrinth seal system (fig. 2(b)) has higher operating temperature and speed capability than face seal systems. However, as the turbine cooling gas pressure (P_1) increases, the leakage has a significant affect on efficiency. Conventional face seal technology can be used in replacing the seal system of figure 2(b) with a single face seal (fig. 2(c)) up to pressures of 125 pounds per square inch (86 N/cm^2), to temperatures of 800° F (700 K), and to sliding speeds of 350 feet per second (107 m/sec) (ref. 3). Seal studies reported (ref. 3) showed that operation at higher pressures, temperatures, or sliding speeds was unsatisfactory from a wear and leakage standpoint; and subsequent analysis revealed that the seal limitations were primarily due to thermal deformations that induced seal imbalance, which, in turn, caused high wear and leakage.

When the turbine cooling gas pressure and temperature exceeds the capability of a conventional face seal (fig. 2(c)), the seal system shown of figure 2(b) can be modified to include a face sump seal (fig. 2(d)) instead of a labyrinth sump seal. Thus, gas leakage into the sump and associated air borne dirt and water is minimized. As in the system of figure 2(b), the leakage of the high pressure (P_1), high-temperature turbine cooling gas is restricted by a labyrinth seal. It should be kept in mind that the turbine cooling gas leakage through this labyrinth seal significantly affects efficiency when the pressure is high. In addition, the conventional face sump seal is speed limited; applications above 400 feet per second (122 m/sec) are beyond current state of the art. Therefore, to meet the needs of advanced engines, improvements in seal speed capability is definitely needed; improvement in temperature and pressure capability is also desired from an efficiency standpoint.

Advanced Seal Requirements

Large-size, high-pressure ratio advanced engines will have compressor discharge pressures in the range of 500 pounds per square inch (345 N/cm^2) and temperatures to 1200° F (922 K). The seals for these engines will have sliding speeds to 500 feet per second (152 m/sec). Engines for supersonic aircraft may generally have lower compres-

sor discharge pressures, but the turbine cooling gas temperatures will also be near 1200° F (922 K). The high sliding speeds in these advanced engines dictate that the sealing surfaces should not be in rubbing contact. Thus, the gas film riding seals such as those under study in NASA Contract NAS3-7609 (ref. 3) are potentially useful in the advanced engines. (Preliminary leakage data for these types of seals are given in ref. 6.) In particular, face seals incorporating self-acting pads for lift augmentation have shown good potential. This seal essentially consist of a conventional seal face in which small self-acting pads are incorporated to augment lift (seal face separation).

An evaluation of the pressure gradient at the sliding interface of a conventional face seal will make apparent the beneficial effect of the self-acting lift pads.

Problem Area in Conventional Face Seals

In the conventional face seal, the pressure gradient between the sealing faces, is affected by the nature of the leakage flow (laminar, turbulent, and/or choked) and by the face geometry (convergent, divergent, or parallel leakage gap). The leakage gap (sealing faces) of a typical face seal presents a long narrow slot to the leakage flow. For instance, a typical sealing gap is 0.0005 inch (0.00127 cm) in height (h) and 0.050 inch (0.127 cm) long (l). Thus the h/l ratio is in the order of 1:100 and viscous forces are important. In order to establish a point of reference, a parallel gap model is considered first. Since the leakage flow is compressible and nearly isothermal, the pressure gradient in the gap has a parabolic shape for slow viscous flow (see fig. 3(a)) and the pressure at any radius (r) in the sealing dam is given by reference 7 as:

$$P(r) = \sqrt{P_1^2 - \frac{24WRT\mu(r - r_i)}{gh^3b}}$$

where:

- P(r) pressure at radius (r), psi
- P₁ sealed pressure, psi
- W weight flow, lb/sec
- R universal gas constant, in.²/(sec²)(K)
- T absolute temperature, K
- μ absolute viscosity, (lb-sec)/in.²
- r radius, in.

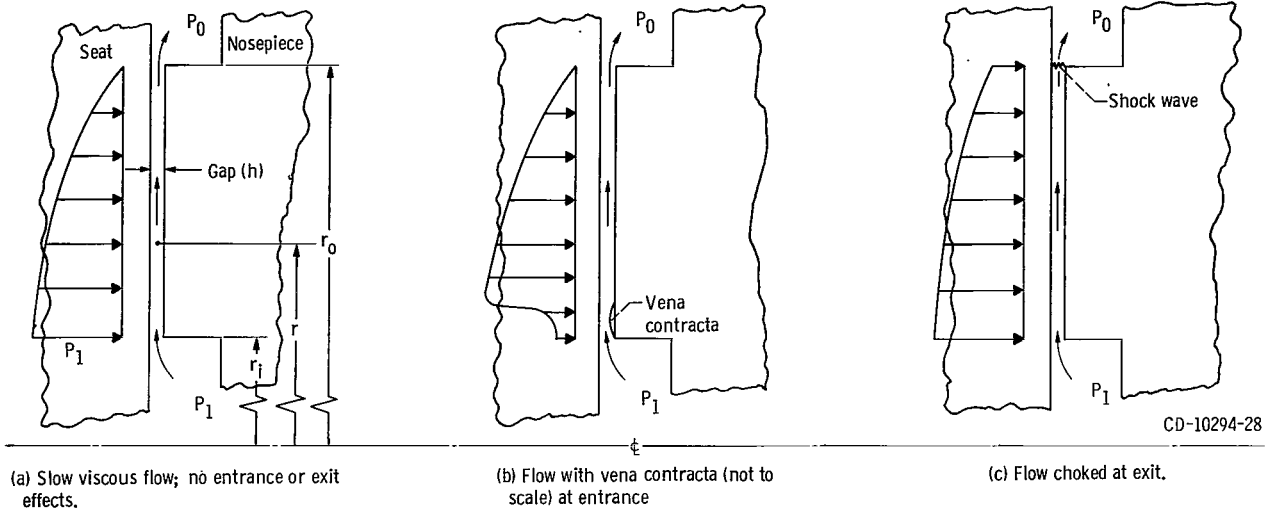


Figure 3. - Pressure gradients for flow in narrow channel.

- r_i inside radius, in.
 r_o outside radius, in.
 g gravitational acceleration, in./sec²
 h gap height, in.
 b width of flow passage = $\pi(r_o + r_i)$

(See ref. 7 for development of equations governing flow in a thin passage.) It should be noted that the radial length of the sealing dam ($r_o - r_i$) is small in comparison to the radius (r_i or r_o) therefore, the leakage can be treated as flow in a rectangular slot (neglecting curvature or increase in flow area as the radius increases).

The preceding formula for calculation of pressure within a narrow passage is valid only for slow flow and it is pointed out that the pressure gradient and leakage mass flow for the reference parallel gap case (fig. 3(a)) can be affected by the following flow conditions:

(1) Vena contracta (fig. 3(b)) - As in the case of an orifice or labyrinth seal, a vena contracta should be expected at the entrance if the sealing face has a sharp edge. A significant reduction in flow area should be expected followed by a small expansion angle. The radial extent of the vena contracta is probably small in comparison to the length of the flow path, hence, its influence on pressure profiles is probably small.

(2) Turbulence - As in the case of pipe flow, turbulence will cause an apparent increase in fluid viscosity, hence affecting the leakage mass flow. The effect of turbulence on pressure profiles was not found in the literature and analysis is needed in this area.

TABLE I. - SEAL PRESSURE, TEMPERATURE, AND SPEED VARIATIONS DURING
FLIGHT FOR A SUPERSONIC MISSION

| | Sealed pressure, absolute | | Pressure ratio across sump seal, P_1/P_0 | Sealed gas temperature | | Seal speed or per cent full engine speed |
|---------|---------------------------|-------------------|--|------------------------|-----|--|
| | psi | N/cm ² | | °F | K | |
| Idle | 50 | 34 | 3 | 550 | 561 | 83 |
| Takeoff | 190 | 131 | 11 | 600 | 589 | 100 |
| Climb | 160 | 110 | 16 | 800 | 700 | 100 |
| Cruise | 215 | 148 | 70 | 1200 | 922 | 100 |
| Descent | 5 | 3.4 | 1.5 | 190 | 361 | 75 |

(3) Choked flow - At high pressure ratios, choking of the flow may occur due to formation of a shock wave at the exit of the leakage gap. Choking will cause an increase in the average gap pressure, thus tending to open the seal gap. Figure 3(c) illustrates the effect of choking on the pressure gradient. It should be noted that the pressure ratio across some seal locations varies markedly during flight (see table I) and choking should be expected.

In addition to the preceeding flow condition, the pressure gradient is affected by the face geometry. The conventional face seal can run without rubbing contact if the sealing faces are convergent in the leakage direction. This convergent gap mode of operation is termed leakage hydrodynamics by some (ref. 8). In principle, the convergent faces operate in the same manner as a stepped face seal or the tapered face seal described in reference 9. Maintenance of the proper amount of convergence from a leakage control standpoint is difficult because of wear and because of the influence of pressure and temperature gradients on face deformation. It should be kept in mind that gas turbine engines subject the seals to a wide variation in pressures and temperatures (e.g., see table I), hence, there exists a wide variation in face deformation that will occur.

The mode of operation of a convergent gap is illustrated in figure 4(a). The solid lines in figure 4(a) depict an equilibrium sealing gap for which the closing force on the nosepiece is in equilibrium with the seal opening force (gap pressure profile in solid lines). If a force causes the gap to close, the increased flow resistance will cause an increase in average gap pressure (dotted line in fig. 4(a)), hence, a restoring force will be built up to return the nosepiece to the equilibrium position. If the gap tends to open beyond the equilibrium position, the gap pressure is reduced and a closing force is built up to restore the nosepiece to the equilibrium position. Thus, the seal can run without rubbing contact. The other consideration in convergent gap analysis is a change in the angle of convergence. If an equilibrium running condition is assumed for a small

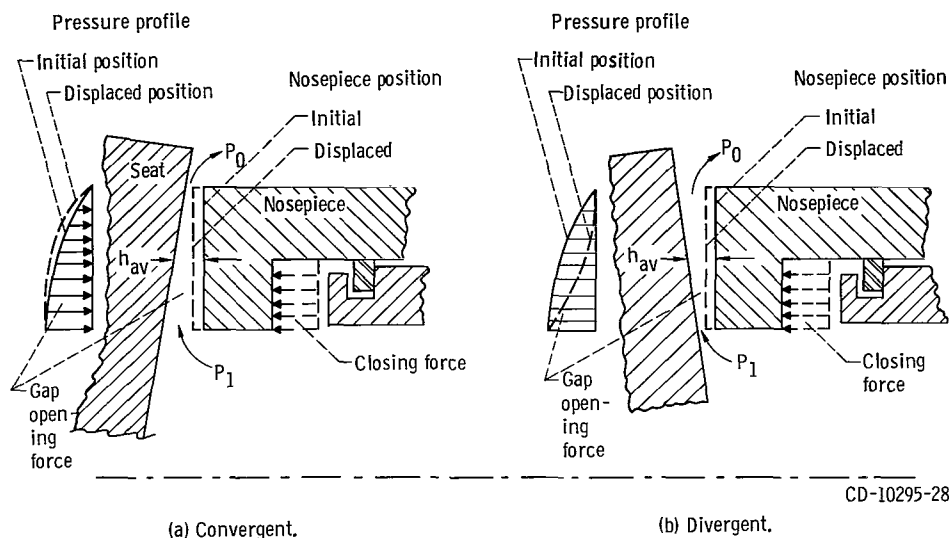


Figure 4. - Sealing face deformation (exaggerated).

angle of convergence, then an increase in convergence angle will increase the average gap pressure. Thus the seal will tend to open more and leakage will increase. A decrease in convergence will cause the opposite effect. As with the parallel face model, entrance effects, turbulent flow, and choked flow must to be considered.

In contrast to the convergent gap, the divergent gap illustrated in figure 4(b), is inherently unstable. A force equilibrium is assumed for the initial nosepiece position (see fig. 4(b)). Now if the gap closes (dotted position), the reduced gap at the inlet, more effectively restricts the leakage flow and, therefore, the average gas pressure decreases to the dotted pressure profile indicated. An unbalanced closing force, therefore, exists and the nosepiece is pushed against the rotating seat. The resulting high-speed rubbing contact results in high wear and the heat generated usually leads to increased face deformation including increased divergence. Divergent deformation is a natural tendency in face sump seals for turbine engines because the hot gas is usually located at the sealing dam inside diameter (the oil-gas mixture is located at the sealing dam outside diameter to preclude oil leakage due to centrifugal force). The temperature gradient in the seal assembly due to the hot gas on one side, and cooler oil on the other side of the seal, tends to cause divergent sealing faces. In addition, shearing of the fluid film within the sealing gap, or rubbing contact if it occurs, heats up the surfaces of the sealing faces and the resulting thermal growth tends to increase the divergence. Therefore, divergent sealing faces are to be expected in many gas turbine seal applications and a method to accommodate these adverse deformations is needed. Small self-acting lift pads incorporated in the face of a conventional seal is one method by which a face seal can be made to operate with positive separation even though the sealing faces are divergent or parallel.

FACE SEAL WITH SELF-ACTING LIFT PADS

The self-acting lift pads consists of a series of shallow recesses arranged circumferentially around the seal under the sealing dam as shown in figure 5. It should be noted that the lift pads are bounded at the inside diameter and outside diameter by the sealed pressure, P_1 . This is accomplished by feed slots connecting the annular groove directly under the sealing dam.

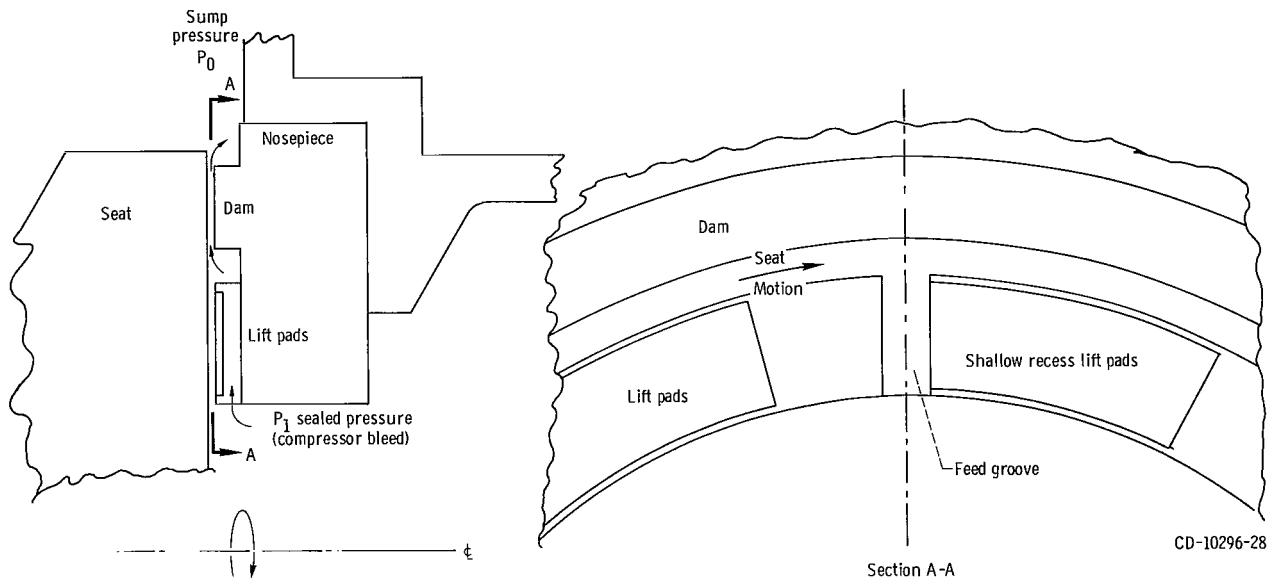
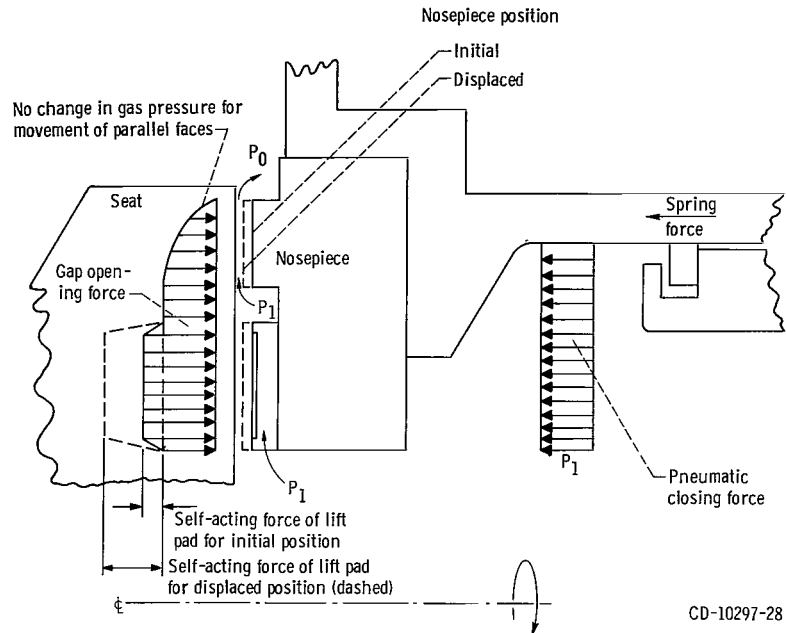


Figure 5. - Face seal with self-acting lift pads.

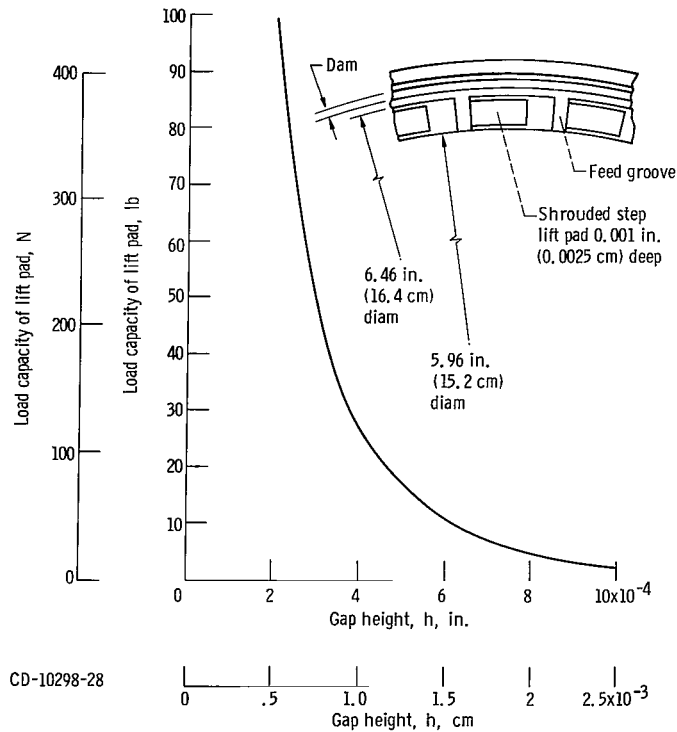
The effect of the self-acting pads on face seal operation is illustrated in figure 6 which shows parallel sealing faces operating without rubbing contact because of a balance between the closing force and the opening force (lift pad forces plus the pressure acting between the sealing faces). For this parallel case, if the seal tends to close, the average pressure in the gap will probably decrease because of increased entrance effects but the gas bearing force increases. Thus a condition of no rubbing contact can prevail except at startup and shutdown.

The self-acting lift pads develop high forces at operating speeds to prevent rubbing contact. For instance, calculations on lift pads (fig. 7) indicate 80-pounds lift-force at 0.0002 inch (0.0005 cm) gap height with 300-pounds-per-square-inch (207-N/cm²) air and 500-feet-per-second (132-m/sec) sliding speed. However, if the gap opens (e.g., to 0.001 in. (0.00254 cm)), then the lift force drops markedly to 4 pounds. Thus there is little tendency for the lift pads to hold the seal open at large gaps where leakage will be



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Figure 6. - Mechanical, pneumatic, and self-acting forces on seal nosepiece.



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Figure 7. - Calculated load capacity of lift pad portion of seal. Number of step pads, 20; absolute pressure 315 psi (217 N/cm²); gas temperature, 1300° F (978 K); sliding speed, 500 feet per second (132 m/sec); parallel sealing faces.

high. The lift pads, therefore, have a force against gap height characteristic (high-gas film stiffness) which makes lift pad incorporation inherently suited to seal operation. This high film stiffness also enhances the capability of the nosepiece to dynamically track the seat face motions.

As mentioned previously, seals with divergent faces, which are a natural tendency in many gas turbine applications, can be made to operate without rubbing contact through incorporation of self-acting lift pads. If the parallel faces of figure 6 become divergent then the average gap pressure should decrease, as in the case of the sequence of figure 4(b), and the gap should tend to close. However, as the gap decreases, the lift-pad force increases rapidly and acts to prevent rubbing contact.

EXPERIMENTAL DATA ON SEALS WITH SELF-ACTING LIFT PADS

Figure 8 is a schematic of the seal and bearing area of a simulated turbine engine sump used to study self-acting seal concepts under Contract NAS3-7609 (ref. 3). The apparatus simulated the roller bearing sump at the turbine location and the structure is typical of engine parts. High-pressure air was introduced at the seal dam inside diameter and air leakage was into the bearing compartment. Seal gas leakage was monitored continuously. Wear measurements of the sealing faces were provided by inspection after running. Other parameters recorded include sealed gas pressure and temperature, rpm, lubricant temperature, and seal nosepiece temperature.

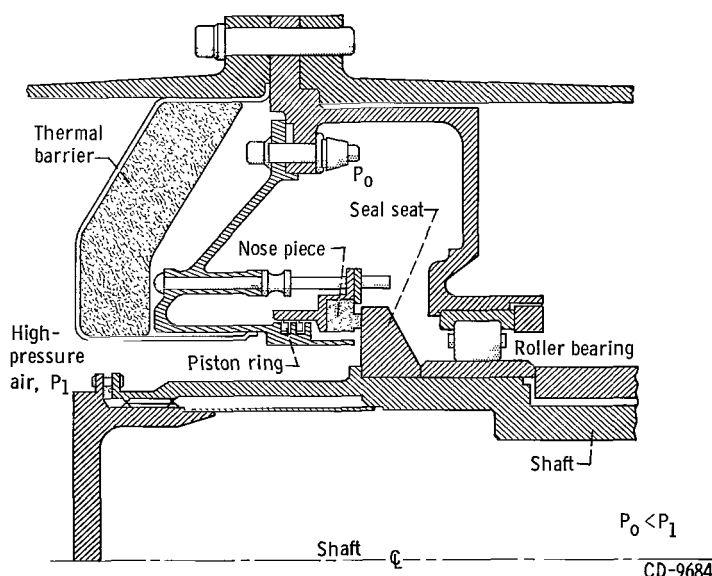


Figure 8. - Bearing and seal assembly in simulated engine sump used in contract NAS 3-7609 (ref. 3) to evaluate face seals with self-acting lift pads.

A typical seal operated in this simulated turbine engine sump was 7.05 inches (17.90 cm) mean diameter. The seal assembly had a rotating seal seat with a tungsten carbide flame plated face and a carbon-graphite nosepiece containing the self-acting lift pads. The sump roller bearing was typical of current design practice and was oil jet lubricated and under-race cooled (not shown). The apparatus was operated to pressures of 200 pounds per square inch (138 N/cm^2), to temperature of 1200° F (922 K), and to speeds of 400 feet per second (122 m/sec).

Details of the seal face developed under Contract NAS3-7609 (ref. 3) at Pratt and Whitney Aircraft Company are given in figure 9. This seal had a bellows secondary seal

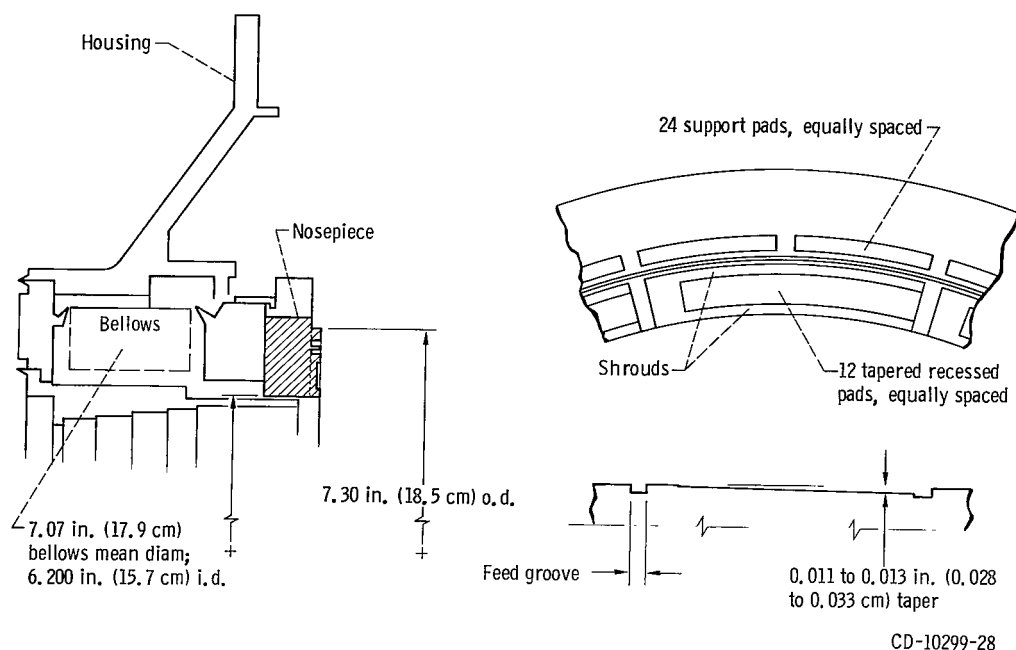


Figure 9. - Face seal with self-acting pads for lift augmentation (ref. 3).

and an oil cooled seat (not shown). Lift augmentation was obtained by means of twelve tapered recesses which act as small self-acting lift pads. Figure 10 shows the seal leakage characteristics as a function of speed when sealing 75° F (297 K) air. At zero rotation speed the leakage rate progressively increased with sealed pressure, reaching 0.00054 pound per second (0.00024 kg/sec) at 200 pounds per square inch (138 N/cm^2). At 200-feet-per-second (61-m/sec) sliding speed and 200 pounds per square inch (138 N/cm^2), the leakage was twice that of zero speed, but still a very low value indicating a small operating gap. At 300 feet per second (91 m/sec) the leakage was only 0.00188 pound per second (0.00085 kg/sec) for 200 pounds per square inch (138 N/cm^2). At 400 feet per second (122 m/sec) the leakage increased sharply, indicating a rub had

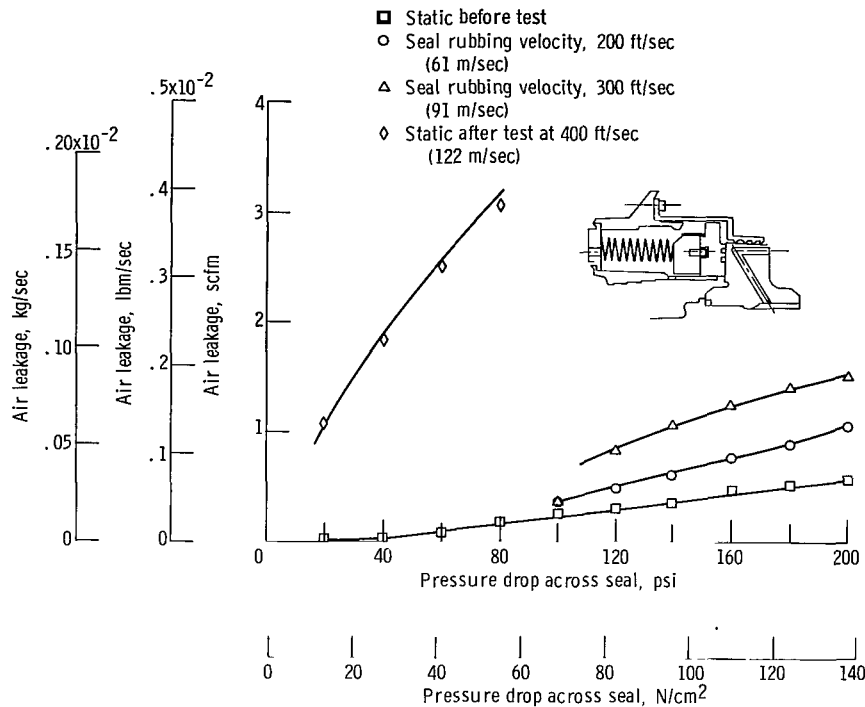


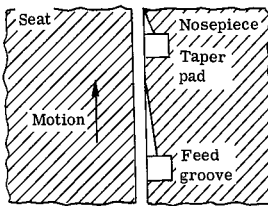
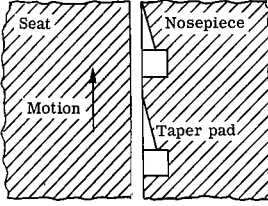
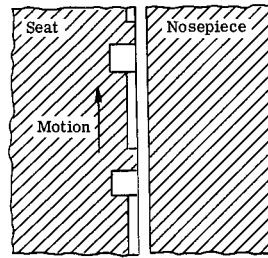
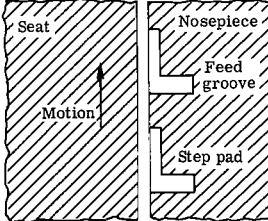
Figure 10. - Leakage as function of pressure and speed for face contact seal with hydrodynamic gas bearing for lift augmentation (from ref. 3).

occurred. Subsequent static tests (see fig. 10) and inspection confirmed that the seal surfaces had been damaged (see table II for pertinent geometric and operating data).

Data obtained under NASA Contract NAS3-12454 show that the self-acting seals, similar to that shown in figure 10, have high temperature capability. In this test series, a seal with a piston ring secondary was used and the face had twenty-four lift pads instead of the twelve pads used in the studies under NASA Contract NAS3-7609. Successful (no rubbing contact) running was obtained in several short term runs with sealed pressures to 175 pounds per square inch (121 N/cm²), temperature to 1100° F (866 K), and sliding speeds of 400 feet per second (122 m/sec). A tabulation of the data are given in table II. The total running time on these seals at various temperatures amounts to about 40 hours. In all cases the seals surfaces were in very good condition after the runs, indicating positive separation of the sealing surfaces had been maintained at operating speeds.

Figure 11 is a schematic of the seal and bearing area of a simulated bearing sump used in Contract NAS3-6267 (ref. 4) to evaluate seal and bearing problems of inerted lubrication systems. The apparatus incorporated a 125-millimeter ball bearing and was operated at 14 000 rpm. The face seal was 6.23-inch (15.80-cm) mean diameter, and was operated at 100-pounds-per-square-inch (69-N/cm²) pressure differential, at sealed

TABLE II. - EXPERIMENTAL DATA ON FACE SEALS WITH THE SELF-ACTING LIFT AUGMENTATION

| Source | | Self-acting lift-pad geometry | | | | | | | Operating data | | | | | | | | Remarks | |
|---------------------------------|-----------|-------------------------------|-------|---------------------|--------------------|--------------------|----------------|--|----------------|---------------------------------|-------------------|-----------------|-----|---------------|-------|--------------------|--------------------|---|
| Contract number or manufacturer | Reference | Mean dam diameter | | Number of lift pads | Lift pad depth | | Lift pad type | Schematic cross section of lift pad | Run time, hr | Sealed pressure, P ₁ | | Gas temperature | | Sliding speed | | Leakage rate, max. | | |
| | | in. | cm | | in. | cm | | | | psi | N/cm ² | °F | K | ft/sec | m/sec | lbm/sec kg/sec | | |
| | | | | | | | | | | | | | | | | | | |
| NAS3-7609 | 3 | 7.05 | 17.90 | 12 | ^a 0.013 | ^a 0.033 | Shrouded taper |  | 3 | 200 | 138 | 75 | 297 | 200 | 61 | 0.00138 | 0.00062 | See figs. 9 and 10 |
| | | | | | | | | | 3 | 200 | 138 | 75 | 297 | 300 | 91 | 0.00188 | 0.00085 | |
| | | | | | | | | | -- | 100 | 69 | 75 | 297 | 400 | 122 | ----- | ----- | Seal face rub, face damaged |
| NAS3-12454 | -- | 7.05 | 17.90 | 24 | ----- | ----- | Shrouded taper |  | 0.5 | 135 | 93 | 75 | 297 | 400 | 122 | 0.00775 | 0.00351 | Surfaces in very good condition after runs |
| | | | | | | | | | 3.5 | 135 | 93 | 1000 | 811 | 400 | 122 | 0.02400 | 0.01089 | |
| | | | | | | | | | 1.5 | 175 | 121 | 1100 | 863 | 400 | 122 | 0.01977 | 0.00895 | |
| NAS3-6267 | 4 | 6.23 | 15.80 | 24 | 0.0005 | 0.00127 | Shrouded step |  | 56 | 100 | 69 | 600 | 589 | 380 | 116 | 0.00863 | 0.00393 | Carbon wear, 0.003 in. (0.0076 cm) |
| | | | | | | | | | 58 | 100 | 69 | 600 | 589 | 380 | 116 | 0.0375 | 0.0170 | Negligible carbon wear |
| | | | | | | | | | 446 | 100 | 69 | 600 | 589 | 380 | 116 | ----- | ----- | ----- |
| NAS3-9428 | 5 | 1.60 | 4.07 | 18 | 0.0002 | 0.0005 | Step |  | 10 | 2.0 | 1.4 | 400 | 478 | 250 | 76 | ----- | ----- | Lift pads outside seal dam |
| | | | | | | | | | 100 | 2.0 | 1.4 | 250 | 394 | 250 | 76 | 7×10 ⁻⁶ | 3×10 ⁻⁶ | Lift pads inside seal dam; negligible wear, 10 starts and stops |
| | | | | | | | | | 353 | 2.0 | 1.4 | 250 | 394 | 250 | 76 | ----- | ----- | Lift pads inside negligible wear, 29 starts and stops |

^aMaximum.

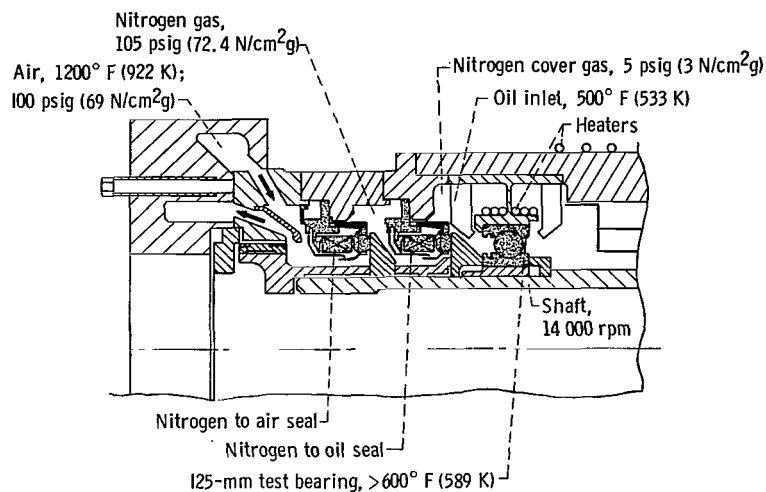


Figure 11. - Bearing and seal assembly in simulated engine sump, used in contract NAS 3-6267 (ref. 4) to evaluate seal and bearing system problems of inerted lubrication systems.

gas temperature of 600° F (589 K), and at a 380-feet-per-second (116-m/sec) sliding speed.

A seal with self-acting pads for lift augmentation (see table II for seal geometry) was used in the simulated turbine engine sump (see fig. 11) under test at the SKF Company under Contract NAS3-6267 (ref. 4). Successful operation (560 hr total) has been obtained with this seal at 100 pounds per square inch (69 N/cm²), pressure differential, 600° F (589 K) gas temperature, and at 380-feet-per-second (116-m/sec) sliding speed. The leakage under these conditions ranged between 0.00863 to 0.03750 pound per second (0.00393 to 0.01700 kg/sec). Wear measurements after the 56 hours of operation, which included eight starts and stops, indicated 0.003 inch (0.0076 cm) carbon wear. After an additional 58 hours, measurements indicated negligible wear. The sealing surfaces after this total of 114 hours of running and 14 starts and stops were in excellent condition.

The Brayton Cycle rotating unit being developed by the Aircsearch Company under Contract NAS3-9428 contains a 1.60-inch (4.07-cm) mean-diameter seal with self-acting lift pads to produce separation of the two sealing surfaces. The seal operates with about a 2.0-pound-per-square-inch (1.4-N/cm²) pressure differential at 250-feet-per-second (76-m/sec) sliding velocity. In the first test series the lift pads were placed above the sealing dam and operation was, in general, satisfactory except there was evidence of some rubbing at the seal dam inside diameter. This rubbing was apparently due to sealing face deformation (thermal), which was greater than the lift produced by the self-acting pads and thus the seal dam inside diameter rubbed the seat. This deformed running mode is illustrated in figure 12(a). By placing the lift pads under the sealing dam (fig. 12(b)) rubbing contact due to the deformation mode of figure 12(a) can be prevented. For the run series with the lift pads under the seal dam, rubbing was

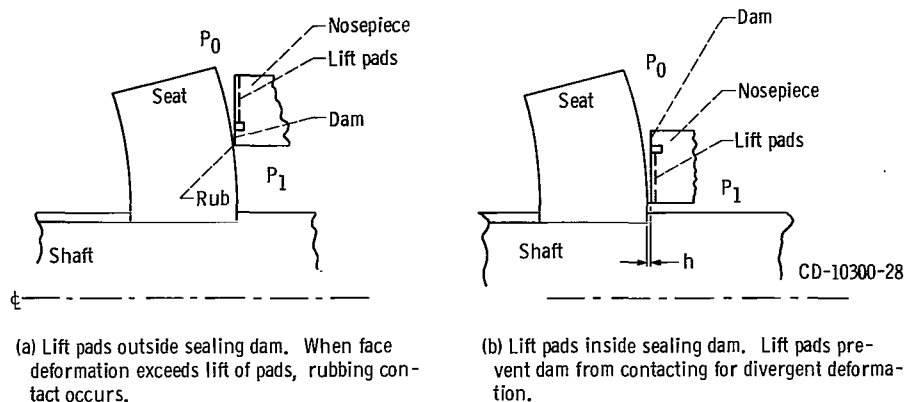


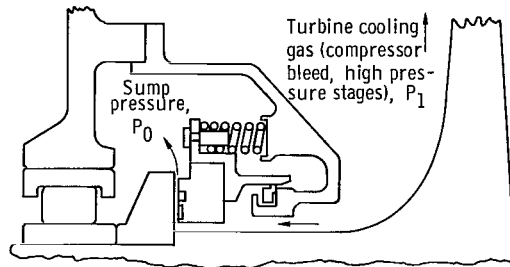
Figure 12. - Effect of lift pad location.

not experienced as indicated by the appearance of the seal faces after the runs, and seal wear was not measurable after 500 hours of operation and twenty-six starts and stops. Table II contains other operational data.

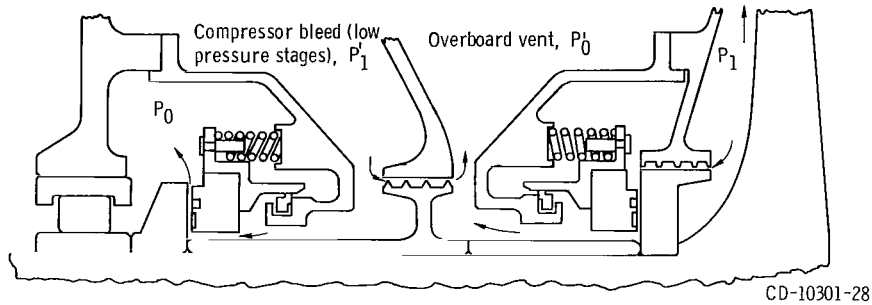
PROPOSED ADVANCED SEAL SYSTEMS

The preceding experimental data and results demonstrate that the face seal with self-acting lift augmentation can operate with positive separation of the sealing faces. Since several tests (table II) demonstrated low leakage potential, this type of face seal is, therefore, a promising candidate for advanced seal systems. One such system, illustrated in figure 13(a), depicts the use of a single sump seal for sealing turbine cooling gas at high pressure and temperature. The projected advantage of such a seal is operation at significantly higher pressures, temperatures, and sliding speeds than now possible with a conventional face seal such as depicted in figure 2(c). The use of a single seal has obvious advantages of minimum weight and space, coupled with low leakage which is important from an efficiency standpoint when pressures are high. However, these advantages must be weighed against fail-safe criteria, such as fire potential in the case of a seal failure.

The fail-safe margin can be improved, and most of the efficiency potential retained by constructing a combined face and labyrinth seal system as depicted in figure 13(b) (no scale intended). A self-acting sump seal allows operation at high speeds and restricts the leakage of moderate temperature and pressure compressor bleed air into the sump. The advantage of this sump seal is low leakage, as compared to a labyrinth seal, hence, associated entrained dirt and water and ducting sizes are minimized. The labyrinth seal can be used in the middle of the seal system to restrict leakage to the over-board vent since the sump pressurization is at moderate pressures (hence, negligible ef-



(a) Single self-acting lift pad seal for sealing turbine cooling gas.



(b) System with two self-acting lift pad seals for sealing turbine cooling gas.

Figure 13. - Proposed seal systems using seals with self-acting pads for lift augmentation (not to scale).

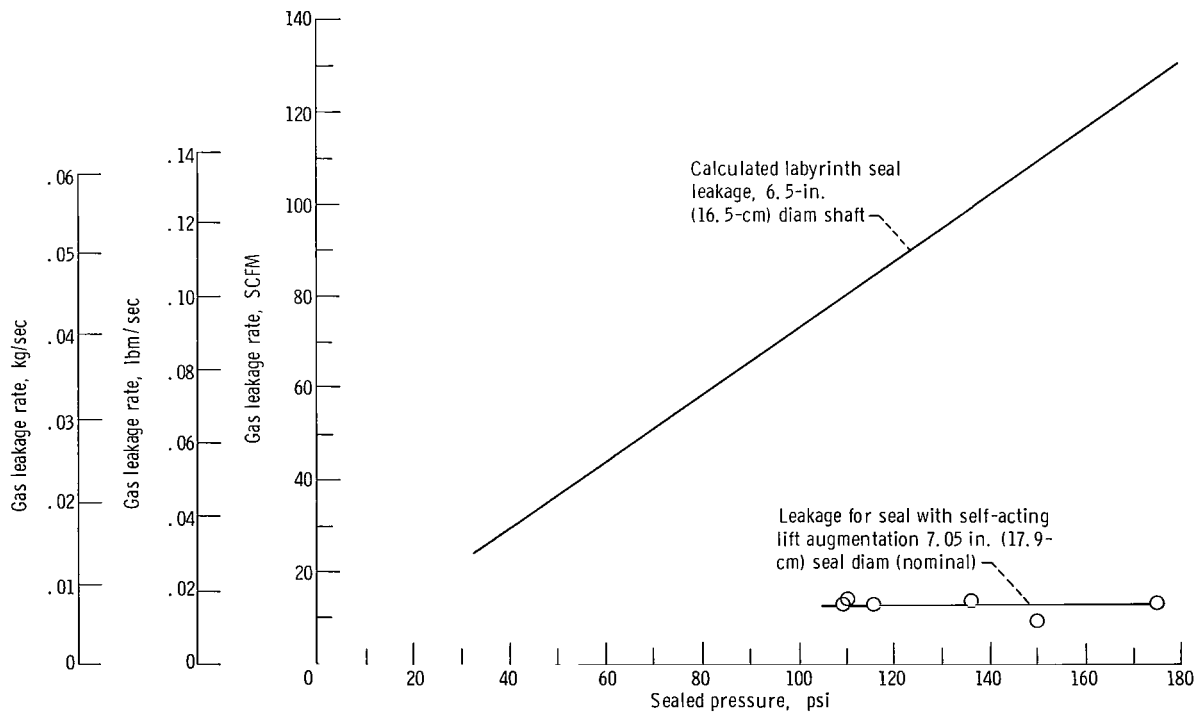


Figure 14. - Gas leakage comparison of labyrinth seal and seal with self-acting lift augmentation. Speed, 400 feet per second (122 m/sec); temperature, 1000° to 1100° F (811 to 863 K).

efficiency penalty). However, a second self-acting seal is used to restrict the leakage of the high-temperature, high-pressure turbine cooling gas to the overboard vent, thus efficiency is improved as compared to a labyrinth system of figures 2(b) and (d). For example, figure 14 shows the calculated leakage for a typical labyrinth seal as compared to experimental data for a seal with self-acting lift pads; in general, the leakage is about ten times that of a face seal with self-acting lift augmentation and this reduction in leakage can be translated into a significant efficiency gain.

CONCLUDING REMARKS

Increased seal operating pressures and temperatures and speeds are required for advanced gas turbine engines. Pressures to 500 pounds per square inch (345 N/cm^2) are to be expected in high-pressure ratio engines. Temperatures will be in the 1200° F (922 K) range and speeds will be as high as 500 feet per second (152 m/sec). To meet these requirements, the seal sliding surfaces should not be in rubbing contact at the operating speeds. Limitations in conventional face seals are due to sealing face deformation which leads to seal force imbalance. In particular, divergent deformation, a natural tendency in many cases, causes rubbing contact and excessive wear. It was demonstrated that a face seal with self-acting pads for lift augmentation can run with positive separation of the sealing surfaces with acceptable leakage rates even through divergence occurs. Studies cited include 6.23-inch (15.80-cm) mean-diameter seal run for 560 hours as well as many short runs on a 7.05-inch (17.90-cm) mean-diameter seal at pressures to 200 pounds per square inch (138 m/sec). The pertinent results are:

1. Leakage rate measurement and inspection of the face after running revealed that the face seal with self-acting pads for lift augmentation can run with positive face separation and maintain acceptable leakage rates. Data and calculations indicated leakage one-tenth of that of a labyrinth seal.

2. The potential of the face seal with self-acting lift pads to operate at higher pressures, temperatures, and sliding speeds than the conventional face seal was demonstrated in short term runs at sealed pressures to 175 pounds per square inch (121 N/cm^2), temperatures to 1100° F (866 K), and sliding speeds to 400 feet per second (122 m/sec). Additional data in an endurance run of 560 hours at 100 pounds per square inch (69 N/cm^2), 600° F (589 K), and 380 feet per second (116 m/sec) confirmed the feasibility of the self-acting type seals.

3. Experimental data indicates high-speed potential, thus the self-acting seal is a candidate for advanced systems in which the speeds are expected to reach 500 feet per

second (152 m/sec). Two advanced seal arrangements using the self-acting seal were described.

Lewis Research Center,
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Cleveland, Ohio, November 27, 1968,
126-15-02-27-22.

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